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Numerical model and investigations of the externally heated valve Joule engine

Jerzy Wojewoda [a,*], Zbyszko Kazimierski [b]

[a] University of Aberdeen, School of Engineering, Fraser Noble Bldg, Aberdeen AB24 3UE, UK
[b] Technical University of Łódź, Institute of Turbomachinery, 219/223 Wólczańska Str., 93-005 Łódź, Poland

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ABSTRACT

The mineral fuels used recently, i.e., oil and gas, will be soon exploited out. This paper presents an idea of the engine where any fuel or solar heat can be used as a source of energy. The proposed model is an externally heated, 2-stroke, valve engine (EHVE). This is a piston-type engine, entirely different from the well-known Stirling one, which is the best known example of such a solution. It works in a closed Joule cycle and is designed to produce a moderate amount of energy. The engine is composed of typical parts met in piston designs: an expander, a compressor, a heater, a cooler and, additionally, two recirculation blowers, which consume a small amount of produced power. An additional advantage is its working medium, which may be simply atmospheric air and the engine has a conventional crankshaft and an oil lubrication system. It has already been proven that operation of the EHVE is possible with satisfactory power and efficiency at the output. Comparisons of the EHVE action with and without recirculation blowers are performed.

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1. Introduction

Our earlier publications [1–4] have been devoted to an externally heated two-stroke piston air engine (EHVE), entirely different from the well-known Stirling engine, [9–11]. This paper is also devoted to a 2-stroke type of the EHVE and the major difference between it and the previous propositions refers to the magnitude of heat exchangers. They were small if compared to the volume of the working cylinder ($V_H/V_E < 1$) and now are much bigger – $V_H/V_E = 3$–5. Additionally, two centrifugal blowers (not present in [1–4]) and a physical model of the complete engine have been added. The blower actions are included in the calculations and are assumed to consume rather a small amount of power, if compared to that produced by the engine. The power consumed by the centrifugal blowers increases monotonously with their dimensions and we assume that it is limited to 10% of the power produced by the engine, but the flow in the exchangers is turbulent. A kind of optimization of power consumed by blowers can be found in Fig. 9.

An important advantage of the proposition is use of the easiest available working medium – atmospheric air and a possibility to apply any source of heat, i.e., any fuel including solar heat or nuclear power. As most mineral fuels used recently, i.e., oil and gas, will be exploited out soon, the idea seems to be worth further development. The EHVE is designated to produce a moderate amount of energy on the basis of an external source of heat. It consists of a few essential parts: an expander, a compressor, a heater, a cooler, two recirculation blowers and works in a closed Joule cycle. The novelty proposed in the paper is an application of two blowers working in loops of high and low pressures, which are linked to the engine heater and the cooler. The air circulation has to be ensured in the exchangers when the valves of the EHVE are closed and the blowers exist. The engine [8], its expander and compressor, are similar in design to those presented in [1–4], though our solutions have been developed independently. Behaviour of heat exchangers used in the experiment [8] was not described, though their action is crucial for the engine operation.

Brzeski and Kazimierski [5] present results of the experimental investigation of the EHVE prototype realizing the idea described in [1–4]. Two heaters of small volumes were employed. The heater contacts the expander and the compressor only during 1/2 period of the engine cycle. During the remaining time, isochoric heating of air enclosed in the heaters should take place. The experiments show, however, that such a type of heating is insufficient for proper operation of the EHVE. The working gas must be changed. Any gas which takes heat by radiation should be applied, and the wall temperature of the heater has to be high enough. Therefore, these changes in the present paper have been introduced.

The issue of the 4-stroke design of the EHVE should be considered thoroughly. The problem is connected with the time when a valve is opening. In 2-stroke engines, the time corresponds to an angle of about 90°, but in the case of 4-stroke engines the camshaft rotates twice as slow. All valves in the 4-stroke engine

* Corresponding author. Tel.: +44 (0) 1224 274176; fax: +44 (0) 1224 272497.
E-mail address: J.Wojewoda@abdn.ac.uk (J. Wojewoda).

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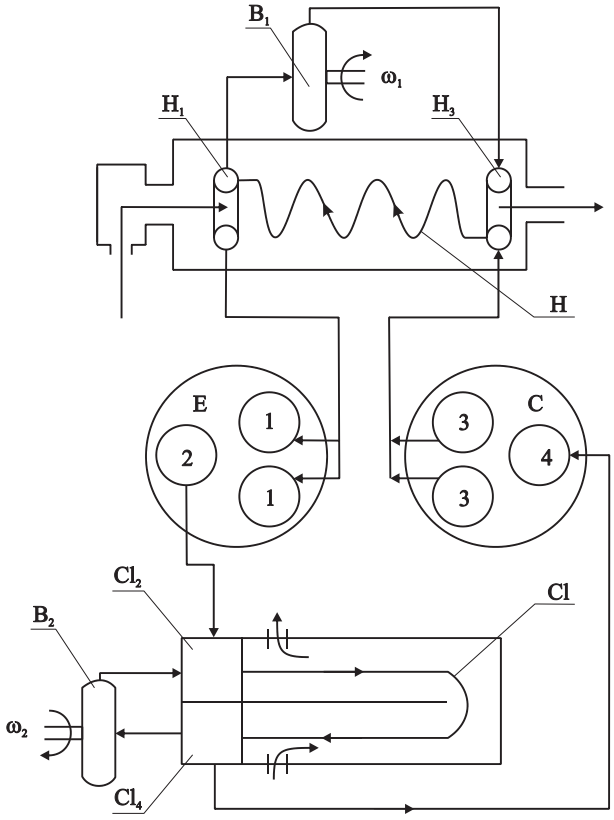

Fig. 1. Scheme of the transversal cross-section of the engine, E – expander, C – compressor, H – heater, CL – cooler, B1, B2 – blowers, 1, 2, 3, 4 – valves.

should be composed of two typical valves like those presented in [6] and such a solution is very difficult in practical realization. The experimental investigation of the 4-stroke EHVE was performed in 2003 and the results were not encouraging enough and the main reason was a very complicated structure of the valves. The presented return to the idea of the 2-stroke engine with a usual value of volumes of heat exchangers and blowers is a way of development of the EHVE when no radiation is taken into account.

The paper is organised as follows. Section 2 presents a principle of the engine operation, a complete theoretical model is described in Section 3, results of computer simulation are presented in Section 4, and finally conclusions are shown in Section 5.

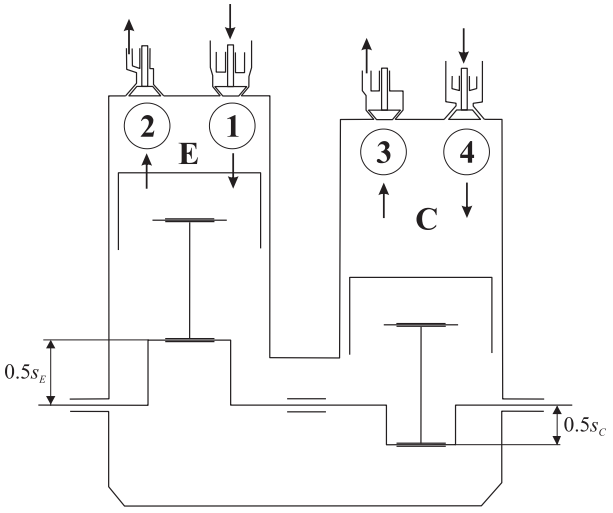

Fig. 2. Scheme of the longitudinal cross-section of the engine.

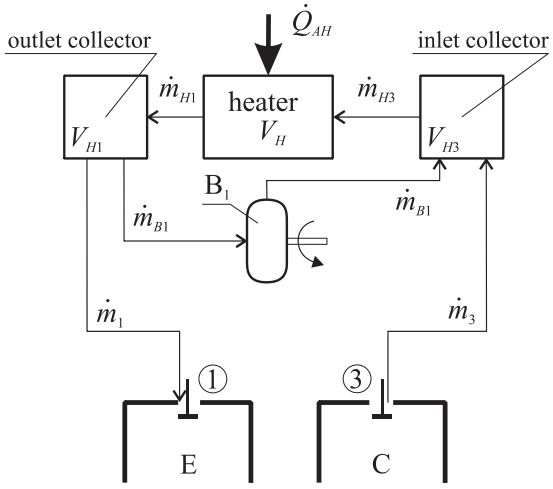

Fig. 3. Scheme of air flow in the heater loop.

2. Principle of the engine operation

The described EHVE represents a 2-stroke-type engine. It is equipped with two cylinders, each of them playing a different role: an expander (E) and a compressor (C) situated between two heat exchangers of moderate volumes: a heater (H) and a cooler (Cl). The phase angle between compressor and expander pistons is equal to 180°. Both the heat exchangers are additionally supplied with their internal blowers raising the rate of heat exchange. The engine works in a closed cycle of the Joule type and its working medium is simply air. The schemes of the EHVE are shown in Figs. 1 and 2. A scheme of flows of the working medium, when the blowers exist, is shown in Figs. 3 (for the heater) and Fig. 4 (for the cooler). Figs. 3 and 4 are added for better understanding of the collector volumes. All valves of the engine are operated by a traditional camshaft. Additionally, the EHVE has a conventional crankshaft and an oil lubrication system.

When the piston of the cylinder E starts to move down from its upper position, valve 1 opens and the working medium at high pressure and temperature values flows from the heater H to the

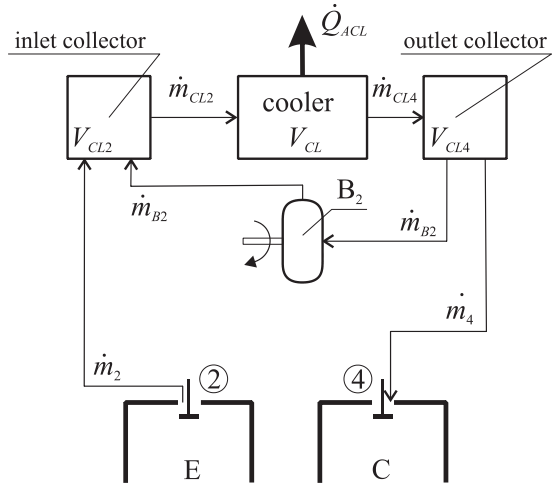

Fig. 4. Scheme of air flow in the cooler loop.

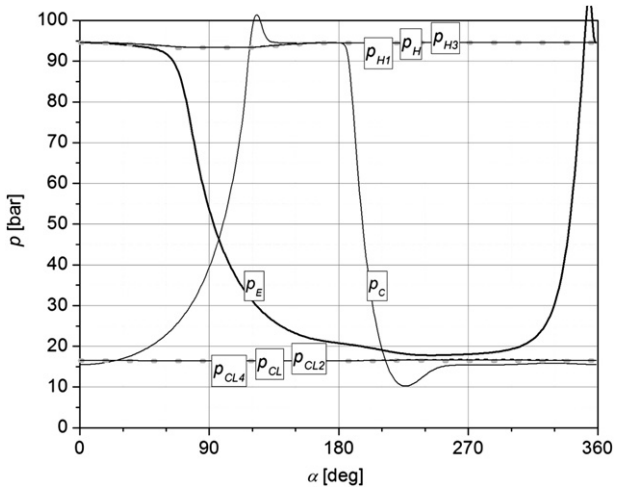

Fig. 5. Pressures in essential parts of the engine at 3000 rpm as a function of the crankshaft angle.

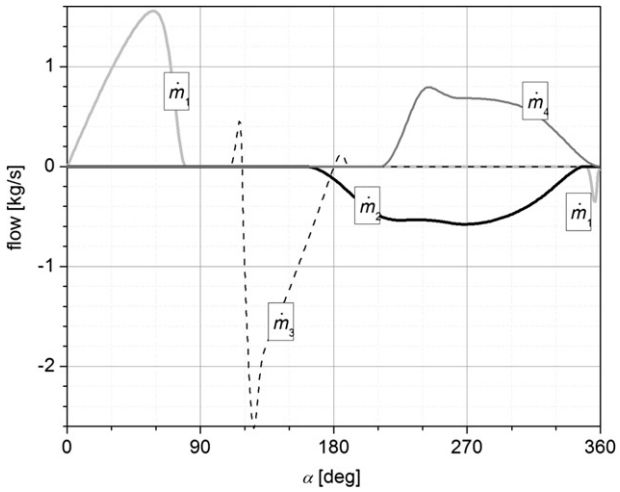

Fig. 7. Mass flow rates of the engine valves for 3000 rpm as a function of the crankshaft angle.

expander E – Figs. 1 and 2. Valve 1 is doubled (it means, its action is realized with 2 identical valves) for better filling of the cylinder E. Both valves 1 close at the angle equal to α_{1c} and further piston movement causes an expansion of the hot air in the locked volume of the cylinder E until the piston reaches its lowest position. At the same time the air locked in the volume of the cylinder C (all the valves closed, see Figs. 1 and 2) is compressed and prepared for flowing to the heater.

Valve 3 opens at about 100° and the compressed air flows into the heater under high pressure – Figs. 1 and 2. In this design, two of valves 3, shown in Fig. 1, are employed for better transportation of the compressed air to the heater H. When the piston C reaches its upper position, the supply of the compressed air to the heater finishes (Fig. 1), and valves 3 close. Then, the pistons in the cylinders E and C start to move again. The piston in the cylinder E moves up and valve 2 opens, allowing the cooler to take the overworked air (Fig. 2). At the same time the piston in the cylinder C moves down, valve 4 opens and the cold air from the cooler flows to the cylinder C. Finally, the pistons reach their primary positions, valves 2 and 4 close and the two-stroke engine cycle ends, realizing a full Joule cycle.

The direction of opening of valves 1 and 3 (Figs. 1 and 2) is inverse than valves 2 and 4, which act in accordance with valves usually met in internal combustion engines.

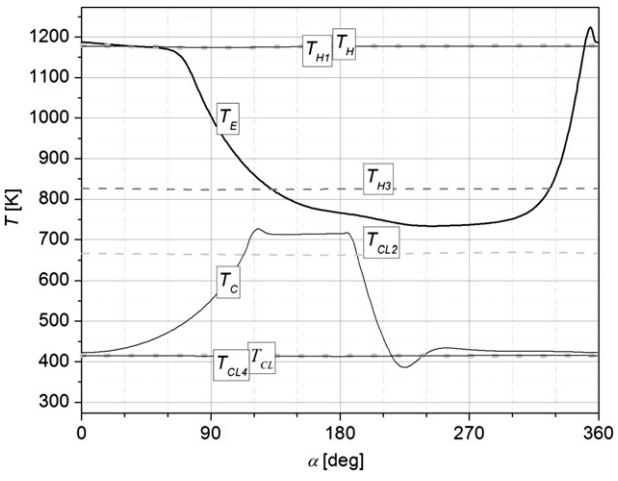

Fig. 6. Temperatures at 3000 rpm – explained at the end of Section 4 as a function of the crankshaft angle.

Publications [1–4] and [8] present a compressor and an expander combined into a single device, which yields a compact cylinder set, however it is difficult in practical realization although $V_C/V_E = 1$. Such a solution is also possible in the presented case, but it could be even more difficult in practical realization because in that case $V_C/V_E < 1$. Besides, in [3] a thermodynamic cycle of the previous version of the EHVE is described. Two self-acting valves (3 and 4) were used in the above-mentioned propositions.

3. Theoretical model

The model presented below is based on the statement formulated in the introduction. All additional assumptions are quoted later in the paper in places they are introduced. We assume approximately that the engine cylinder (cooled) operates in such a way that a heat exchange inside it does not exist.

The most important for cylinders are the time-dependent relations and this has been taken into account, whereas a dependence on spatial variables has been neglected, so the model is described and simulated in the time domain only.

The heat exchangers are situated between two collectors, see Figs. 3 and 4. The inlet and outlet collectors are excluded from the heat exchangers and it is assumed that no heat streams inside the collectors exist. The collectors play, at the same time, a role of settling chambers, which are important for the behaviour of

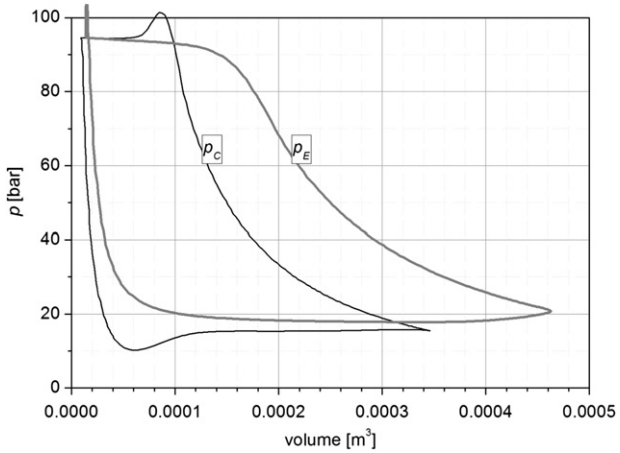

Fig. 8. *p-V* diagram for 3000 rpm.

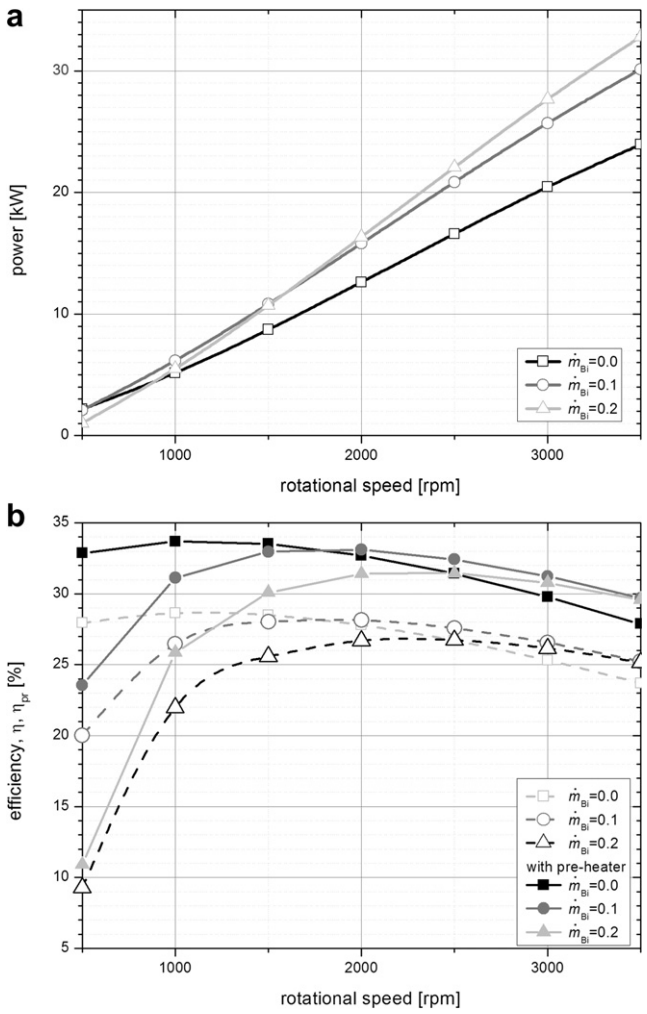

Fig. 9. Power (a) and efficiency (b) of the engine as a function of its rotational speed.

blowers. The preliminary calculations show that pressures and temperatures in the collectors are almost constant (Figs. 5 and 6). Such a situation takes place when the rotation of the engine is constant and a volume of the collector is greater than a volume of the cylinder. This allows for considering the blowers as almost stationary devices. This is explained in more detail in Section 4 of the paper. The power consumed by the blowers will be determined.

In both heat exchangers the spatial changes may be important and they were earlier calculated in [4], as space and time variables, because the temperature of the working air inside the heater in fact grows from the inlet to outlet collectors – Figs. 1 and 3. A simpler approach to the problem is also possible and it is applied in this paper.

The heat exchangers can be described as follows. The proper heater and cooler are counter-current heat exchangers, see Fig. 1, where a temperature drop remains almost constant along their length, [12]. The heat exchangers are treated as time-dependent only, assuming that all important changes are the same along the overall length of the exchangers and considering each of them in a single plane only. The plane is situated at the outlet of the exchanger tubes. The approximate approach used here allows for applying maximal and minimal temperatures of the exchanger walls in the discussed case. The heat volumes delivered to and subtracted from the EHVE in the heat exchangers are estimated accordingly.

The applied approach to the exchangers is sufficient for calculations of the benefits resulting from using the blowers in the installation. The accuracy of the approach is checked many times during the computer simulation in Section 4.

Such a treatment of the parameters is of course a kind of approximation, but it allows us to use a moderate number of equations in case of a time-dependent model. These are non-linear ordinary, differential equations of mass and energy conservation, supplemented by some non-linear algebraic ones. The details of the theoretical model will be explained later in the paper.

Firstly, let us consider geometry of cylinders and valves, then mass flow rates through the EHVE valves will be presented and, finally, essential differential equations for the main elements of the engine will be discussed.

3.1. Geometry of cylinder volumes and the valve cross-section area

The volume of the expander is described by:

$$V_E = A_E\left[h_{0E} + \frac{s_E}{2}(1-\cos\omega t)\right], \tag{1}$$

where: $A_E = \pi d_E^2/4$, and its derivative is:

$$\frac{dV_E}{dt} = A_E\frac{s_E\omega}{2}\sin\omega t. \tag{2}$$

The volume of the compressor equals to:

$$V_C = A_C\left[h_{0C} + \frac{s_C}{2}(1+\cos\omega t)\right], \tag{3}$$

where: $A_C = \pi d_C^2/4$, and its derivative is:

$$\frac{dV_C}{dt} = -A_C\frac{s_C\omega}{2}\sin\omega t. \tag{4}$$

The cross-section areas of the valves – A_j ($j = 1, 2, 3, 4$) – are described by the following formulae (where $\alpha = \omega t$):

1) for $\alpha_{j,0} \le \alpha \le \alpha_{j,0} + \Delta\alpha_j$:

$$A_j(\alpha) = \frac{1}{2}A_{j,\max}\left(1-\cos\pi\frac{\alpha-\alpha_{j,0}}{\Delta\alpha_j}\right); \tag{5}$$

2) for $\alpha_{j,0} + \Delta\alpha_j \le \alpha \le \alpha_{j,c} - \Delta\alpha_j$:

$$A_j(\alpha) = A_{j,\max}; \tag{6}$$

3) for $\alpha_{j,c} - \Delta\alpha_j \le \alpha \le \alpha_{j,c}$:

$$A_j(\alpha) = \frac{1}{2}A_{j,\max}\left(1-\cos\pi\frac{\alpha-\alpha_{j,c}}{\Delta\alpha_j}\right), \tag{7}$$

and:

$$A_{j,\max} = \pi d_j \bullet h_j \bullet \sin 45^o. \tag{8}$$

All the data: $\alpha_{j,0}$, $\alpha_{j,c}$, $\Delta\alpha_j$, d_j, h_j are assumed in the computer program. The last term in Eq. (8) results from a typical design of the shape of valves.

3.2. Mass flow rates through the engine valves

Mass flow rates are calculated using Bernoulli and continuity equations for gas, where the losses are determined by coefficients ζ_j. The general form of this equation (for $j = 1, 2, 3, 4$) is:

$$\dot{m}_j = \frac{\zeta_j A_j p_b}{\sqrt{RT_b}}\left(\frac{p_a}{p_b}\right)^{\frac{1}{\kappa}}\sqrt{\frac{2\kappa}{\kappa-1}\left[1-\left(\frac{p_a}{p_b}\right)^{\frac{\kappa-1}{\kappa}}\right]}, \tag{9}$$

and, if:

$$\frac{p_a}{p_b} \le \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}}, \tag{10}$$

then:

$$\dot{m}_j = \frac{\zeta_j A_j p_b \sqrt{\kappa}}{\sqrt{RT_b}}\left(\frac{2}{\kappa+1}\right)^{\frac{\kappa+1}{2(\kappa-1)}}. \tag{11}$$

Values of the ζ_j coefficients equal to 0.85 are usually used in similar designs. These values are found in literature; see for example [14]. Pressures p_a and p_b and temperatures T_a, T_b in Eqs. (9)–(11) are calculated as follows:

$$\begin{aligned}
&\text{for } j = 1\\
&\quad \text{if} \quad p_{H1} > p_E : p_b = p_{H1},\ T_b = T_{H1},\ p_a = p_E,\\
&\quad \text{if} \quad p_E > p_{H1} : p_b = p_E,\ T_b = T_E,\ p_a = p_{H1},\\
&\text{for } j = 2\\
&\quad \text{if} \quad p_E > p_{Cl2} : p_b = p_E,\ T_b = T_E,\ p_a = p_{Cl2},\\
&\quad \text{if} \quad p_{Cl2} > p_E : p_b = p_{Cl2},\ T_b = T_{Cl2},\ p_a = p_E,\\
&\text{for } j = 3\\
&\quad \text{if} \quad p_C > p_{H3} : p_b = p_C,\ T_b = T_C,\ p_a = p_{H3},\\
&\quad \text{if} \quad p_{H3} > p_C : p_b = p_{H3},\ T_b = T_{H3},\ p_a = p_C,\\
&\text{for } j = 4\\
&\quad \text{if} \quad p_C > p_{Cl4} : p_b = p_C,\ T_b = T_C,\ p_a = p_{Cl4},\\
&\quad \text{if} \quad p_{Cl4} > p_C : p_b = p_{Cl4},\ T_b = T_{Cl4},\ p_a = p_C,
\end{aligned} \tag{12}$$

The mass streams $\dot{m}_{H1}$ and $\dot{m}_{H3}$ between parts of the heater will be discussed afterwards.

3.3. Essential equations for EHVE parts

Let us consider the expander first. In its volume V_E, the equation of mass conservation has a form:

$$\frac{d\rho_E}{dt} = \frac{1}{V_E}\left(\pm\dot{m}_1 \pm \dot{m}_2 - \rho_E\frac{dV_E}{dt}\right). \tag{13}$$

The signs of the mass streams $\dot{m}_1$ and $\dot{m}_2$ are determined as follows:

$$\begin{aligned}
\dot{m}_1 &\begin{cases} \text{if} & p_{H1} > p_E, \quad \text{sign} \quad ''+'', \\ \text{if} & p_E > p_{H1}, \quad \text{sign} \quad ''-''. \end{cases}\\
\dot{m}_2 &\begin{cases} \text{if} & p_E > p_{Cl2}, \quad \text{sign} \quad ''-'', \\ \text{if} & p_{Cl2} > p_E, \quad \text{sign} \quad ''+''. \end{cases}
\end{aligned} \tag{14}$$

The equation of energy conservation for the expander takes a form:

$$\frac{dT_E}{dt} = \frac{1}{\rho_E V_E}\left[-T_E(\pm\dot{m}_1 \pm \dot{m}_2) + \kappa\left(\begin{Bmatrix} +\dot{m}_1 T_{H1} \\ -\dot{m}_1 T_E \end{Bmatrix} + \begin{Bmatrix} +\dot{m}_2 T_{Cl2} \\ -\dot{m}_2 T_E \end{Bmatrix}\right) - \frac{p_E}{c_v}\frac{dV_E}{dt}\right] \tag{15}$$

The signs of the mass stream terms $\dot{m}_1$ and $\dot{m}_2$ can be determined from Eq. (14). The temperature in the parenthesis is chosen accordingly to the relations expressing the mass flows. Such a notation is repeated in the following equations of energy conservation. The pressure p_E inside the expander is calculated from a typical equation of state for ideal gas:

$$p_E = \rho_E T_E R. \tag{16}$$

If the equation of state for real gas (e.g., Peng-Robinson) is taken into account, the error for the extreme parameters of air (100 bar and 1000 °C) is less than 2%. So, we remain with the quoted equation.

In case of the compressor, we have similar dependencies. Its equation of mass conservation takes a form:

$$\frac{d\rho_C}{dt} = \frac{1}{V_C}\left(\pm\dot{m}_3 \pm \dot{m}_4 - \rho_C\frac{dV_C}{dt}\right) \tag{17}$$

The signs of the mass streams $\dot{m}_3$ and $\dot{m}_4$ are given below:

$$\begin{aligned}
\dot{m}_3 &\begin{cases} \text{if} & p_C > p_{H3}, \quad \text{sign} \quad ''-'', \\ \text{if} & p_{H3} > \text{pc}, \quad \text{sign} \quad ''+''. \end{cases}\\
\dot{m}_4 &\begin{cases} \text{if} & p_C > p_{Cl4}, \quad \text{sign} \quad ''-'', \\ \text{if} & p_{Cl4} > p_C, \quad \text{sign} \quad ''+''. \end{cases}
\end{aligned} \tag{18}$$

The equation of energy conservation for the compressor is as follows:

$$\frac{dT_C}{dt} = \frac{1}{\rho_C V_C}\left[-T_C(\pm\dot{m}_3 \pm \dot{m}_4) + \kappa\left(\begin{Bmatrix} +\dot{m}_3 T_{H3} \\ -\dot{m}_3 T_C \end{Bmatrix} + \begin{Bmatrix} +\dot{m}_4 T_{Cl_4} \\ -\dot{m}_4 T_C \end{Bmatrix}\right) - \frac{p_C}{c_v}\frac{dV_C}{dt}\right] \tag{19}$$

Eqs. (15) and (19) assume that no heat exchange takes place in the cylinders of the EHVE. The signs of the mass stream terms $\dot{m}_3$ and $\dot{m}_4$ are determined in Eq. (18). The pressure p_C inside the compressor is calculated from the equation of state:

$$p_C = \rho_C T_C R. \tag{20}$$

Let us consider the heater model now. It is divided into three parts of constant volumes, namely:

1. inlet collector – V_{H3},
2. proper heater (a counter-current type) – V_H,
3. outlet collector – V_{H1}.

The volume of the proper heater and its surface are determined by:

$$\begin{aligned}
V_H &= \tfrac{\pi d_H^2}{4} l_H n_H,\\
A_H &= \pi d_H l_H n_H.
\end{aligned} \tag{21}$$

The working air flows from the compressor C to the inlet collector V_{H3} through valve 3 (Fig. 3). Then, it flows to the proper heater volume V_H, transfers to the outlet collector V_{H1}, and finally reaches the expander E volume through valve 1 – Fig. 3. The blower B1 transports the air from the outlet collector of the heater to its inlet. We assume that the temperature of the outlet collector is also transferred to the inlet one and the pressure drop between them is maintained by the blower B1. The heater collectors are considered as settling chambers for the flow. The mass flow rates $\dot{m}_{H1}$ and $\dot{m}_{H3}$ are usually not equal to each other and their calculations are performed using Eq. (9), with the following relations of pressures and temperatures:

$$\begin{aligned}
&\text{for } j = \text{H3}:\\
&\quad \text{if} \quad p_{H3} > p_H : p_b = p_{H3};\quad T_b = T_{H3};\ p_a = p_H,\\
&\quad \text{if} \quad p_H > p_{H3} : p_b = p_H;\quad T_b = T_H;\ p_a = p_{H3},\\
&\text{for } j = \text{H1}:\\
&\quad \text{if} \quad p_H > p_{H1} : p_b = p_H;\quad T_b = T_H;\ p_a = p_{H1},\\
&\quad \text{if} \quad p_{H1} > p_H : p_b = p_{H1};\quad T_b = T_{H1};\ p_a = p_H.
\end{aligned} \tag{22}$$

The cross-sections, necessary for Eq. (9), are assumed as:

$$A_{H1} = A_{H3} = \frac{\pi d_H^2}{4} n_H, \tag{23}$$

and $\zeta_{H1} = \zeta_{H3} = 0.85$. The signs of the mass streams $\dot{m}_{H1}$ and $\dot{m}_{H3}$ will be defined later.

The blower in the loop of hot air circulation assures that the Reynolds number calculated for the exchanger tubes is not less than 10^4, when valves 1 and 3 remain closed. The Reynolds number caused by a flow from the first blower in such a situation is:

$$\mathrm{Re}_H = 4\dot{m}_{B1}/(\pi d_H \mu_H n_H), \tag{24}$$

where $\dot{m}_{B1}$ is calculated according to above-mentioned condition $\mathrm{Re}_H = 10^4$.

The Nusselt number for the proper heater tubes is calculated from the formula:

$$\mathrm{Nu}_H = 0.022(\mathrm{Re}_H)^{0.8}(\mathrm{Pr}_H)^{0.6}. \tag{25}$$

where Re_H is calculated as a varying value and uses $\dot{m}_H$ flowing through the heater. The value $\dot{m}_{B1}$ is a small part of $\dot{m}_H$, but exists when valves 1 and 3 are closed and the blower B1 works in the heater loop, Fig. 3.

The coefficient of heat transfer equals to:

$$\alpha_{AH} = \mathrm{Nu}_H \lambda_H / d_H. \tag{26}$$

The heat stream delivered in the complete heater is a function of time and is determined as follows:

$$\dot{Q}_{AH} = \alpha_{AH} A_H (T_{wH} - T_H), \tag{27}$$

where T_{wH} – according to the described theoretical model – is the maximum temperature of the heater wall for a given rotational speed of the engine, and T_H is a time-dependent temperature of the air inside heater tubes, at their outlet position.

The pressures in both collectors are calculated in the following way. In the inlet, boosted collector H3, the pressure increases by Δp_{H3}, see – Eq. (31). The proper heater is composed of long tubes, which caused the pressure friction losses Δp_{Hf}, calculated for the overall length of tubes. Therefore, the pressure in the outlet collector H1 decreases by Δp_{Hf} and is given in Eq. (39).

Now, the essential equations for the heater will be presented.

The calculation for the inlet collector V_{H3} is based on the essential equations for this part of the heater. The equation of mass conservation takes a form:

$$\frac{\mathrm{d}\rho_{H3}}{\mathrm{d}t} = \frac{1}{V_{H3}}(\pm\dot{m}_3 \pm \dot{m}_{H3} + \dot{m}_{B1}). \tag{28}$$

The signs of the mass streams are:

$$\dot{m}_3 \begin{cases} \text{if} & p_C > p_{H3}, \quad \text{sign} \quad ''+'', \\ \text{if} & p_{H3} > p_C, \quad \text{sign} \quad ''-''. \end{cases}$$
$$\dot{m}_{H3} \begin{cases} \text{if} & p_{H3} > p_H, \quad \text{sign} \quad ''-'', \\ \text{if} & p_H > p_{H3}, \quad \text{sign} \quad ''+''. \end{cases} \tag{29}$$

The equation of energy conservation is calculated from the formula:

$$\frac{\mathrm{d}T_{H3}}{\mathrm{d}t} = \frac{1}{\rho_{H3} V_{H3}}\Big[- T_{H3}(\pm\dot{m}_3 \pm \dot{m}_{H3} + \dot{m}_{B1}) + \kappa\left(\begin{Bmatrix} +\dot{m}_3 T_C \\ -\dot{m}_3 T_{H3} \end{Bmatrix} + \begin{Bmatrix} +\dot{m}_{H3} T_H \\ -\dot{m}_{H3} T_{H3} \end{Bmatrix} + \dot{m}_{B1} T_{H1}\right)\Big] \tag{30}$$

As the action of blowers is unidirectional, the positive mass flow rate of the blower B1 is assumed.

Finally, the equation of the pressure p_{H3} is:

$$p_{H3} = \rho_{H3} T_{H3} R + \Delta p_{H3}. \tag{31}$$

The essential equations for the proper part of the heater V_H are as follows. The equation of mass conservation:

$$\frac{\mathrm{d}\rho_H}{\mathrm{d}t} = \frac{1}{V_H}(\pm\dot{m}_{H1} \pm \dot{m}_{H3}), \tag{32}$$

where:

$$\dot{m}_{H1} \begin{cases} \text{if} & p_{H1} > p_H, \quad \text{sign} \quad ''+'', \\ \text{if} & p_H > p_{H1}, \quad \text{sign} \quad ''-''. \end{cases}$$

and (33)

$$\dot{m}_{H3} \begin{cases} \text{if} & p_H > p_{H3}, \quad \text{sign} \quad ''-'', \\ \text{if} & p_{H3} > p_H, \quad \text{sign} \quad ''+''. \end{cases}$$

The equation of energy conservation has a form:

$$\frac{\mathrm{d}T_H}{\mathrm{d}t} = \frac{1}{\rho_H V_H}\Big[- T_H(\pm\dot{m}_{H1} \pm \dot{m}_{H3}) + \kappa\left(\begin{Bmatrix} +\dot{m}_{H1} T_{H1} \\ -\dot{m}_{H1} T_H \end{Bmatrix} + \begin{Bmatrix} +\dot{m}_{H3} T_{H3} \\ -\dot{m}_{H3} T_H \end{Bmatrix}\right) + \frac{\dot{Q}_{AH}}{c_v}\Big]. \tag{34}$$

The signs of the mass streams $\dot{m}_{H1}$ and $\dot{m}_{H3}$ are given in Eq. (33), but the term $\dot{Q}_{AH}$ is calculated for the complete heater. The pressure at the end of heater tubes is calculated as follows:

$$p_H = \rho_H T_H R. \tag{35}$$

Finally, the outlet collector of volume V_{H1} is discussed. Its equation of mass conservation is:

$$\frac{\mathrm{d}\rho_{H1}}{\mathrm{d}t} = \frac{1}{V_{H1}}(\pm\dot{m}_{H1} \pm \dot{m}_1 - \dot{m}_{B1}), \tag{36}$$

where:

$$\dot{m}_{H1} \begin{cases} \text{if} & p_H > p_{H1}, \quad \text{sign} \quad ''+'', \\ \text{if} & p_{H1} > p_H, \quad \text{sign} \quad ''-''. \end{cases}$$

and (37)

$$\dot{m}_1 \begin{cases} \text{if} & p_{H1} > p_E, \quad \text{sign} \quad ''-'', \\ \text{if} & p_E > p_{H1}, \quad \text{sign} \quad ''+''. \end{cases}$$

The equation of energy conservation takes a form:

$$\frac{\mathrm{d}T_{H1}}{\mathrm{d}t} = \frac{1}{\rho_{H1} V_{H1}}\Big[- T_{H1}(\pm\dot{m}_{H1} \pm \dot{m}_1 - \dot{m}_{B1}) + \kappa\left(\begin{Bmatrix} +\dot{m}_{H1} T_H \\ -\dot{m}_{H1} T_{H1} \end{Bmatrix} + \begin{Bmatrix} +\dot{m}_1 T_E \\ -\dot{m}_1 T_{H1} \end{Bmatrix} - \dot{m}_{B1} T_{H1}\right)\Big]. \tag{38}$$

The signs of the mass stream $\dot{m}_{H1}$ and $\dot{m}_1$ are given in Eq. (37).

The pressure inside the collector volume V_{H1} is calculated from the following equation:

$$p_{H1} = \rho_{H1} T_{H1} R - \Delta p_{Hf}. \tag{39}$$

At last, a cooler of the constant volume is presented. The cooler volume is divided into three parts, in a similar way to the heater:

1. inlet collector $-V_{Cl2}$,
2. proper cooler (a counter-current type) $-V_{Cl}$,
3. outlet collector$-V_{Cl4}$.

The volume of the cooler and the cooling surface are determined by:

$$V_{Cl} = \frac{\pi d_{Cl}^2}{4} l_{Cl} n_{Cl}, \quad A_{Cl} = \pi d_{Cl} l_{Cl} n_{Cl}. \tag{40}$$

The working air is flowing from the expander E to the inlet collector V_{Cl2}, through valve 2 – see Fig. 4. Then it is transferred from the inlet collector through the proper cooler V_{Cl} to the outlet collector V_{Cl4}, finally reaching the volume of the compressor C through valve 4.

The blower B2 forces the working medium to transfer from the outlet collector to the inlet one in the cooler. We assume that the temperature in the outlet collector is transferred directly to the inlet and a pressure drop between them is covered by the B2 blower action. Additionally, the cooler collectors are considered to be settling chambers. The blower, which works in the cooling loop, causes that its Reynolds number is not less than 10^4 when valves 2 and 4 are closed. This number calculated for the cooler tubes being forced by the second blower (when valves 2 and 4 are closed) is determined as:

$$\mathrm{Re}_{Cl} = 4\ \dot{m}_{B2}/(\pi d_{Cl} \mu_{Cl} n_{Cl}), \tag{41}$$

where the value of $\dot{m}_{B2}$ is calculated according to the above equation for values of the $\mathrm{Re}_{Cl} = 10^4$.

The Nusselt number for the proper cooler part tubes is calculated from:

$$\mathrm{Nu}_{Cl} = 0.022(\mathrm{Re}_{Cl})^{0.8}(\mathrm{Pr}_{Cl})^{0.6}, \tag{42}$$

where Re_{Cl} is calculated as a varying values and uses $\dot{m}_{Cl}$ flowing through the cooler. The value $\dot{m}_{B2}$ is a small part of $\dot{m}_{Cl}$, but exists when valves 2 and 4 are closed and the blower B2 works in the cooler loop, Fig. 4.

The coefficient of the heat transfer in the cooler is determined as:

$$\alpha_{ACl} = \mathrm{Nu}_{Cl} \lambda_{Cl} / d_{Cl}. \tag{43}$$

The heat stream in the complete cooler section is a following time-dependent function:

$$\dot{Q}_{ACl} = \alpha_{ACl} A_{Cl} (T_{wCl} - T_{Cl}), \tag{44}$$

where according to the described theoretical model, the temperature T_{wCl} is the minimum temperature of the cooler wall, and T_{Cl} – of the air inside the cooler tubes, at their outlet. The temperature of the air inside the cooler falls, in fact, between values in the collectors Cl2 and Cl4, but the heat stream described by Eq. (44) is calculated for the overall length of its tubes. According to the theoretical model, the temperature is calculated at the outlet from the cooler tubes (the last plane of the proper part of the cooler) and is time-dependent only.

The pressures in the collectors of the cooler are described similarly to the collectors of the heater. The inlet, boosted collector is now Cl2 and the outlet collector is Cl4. Similarly, the pressure increases in Cl2 by Δp_{Cl2}, but decreases in Cl4 by Δp_{Clf}.

The mass streams flowing in and out of the cooler are usually not equal to each other and their calculations are performed using Eq. (9), with the following relations:

$$\begin{aligned}
&\text{for } j = \mathrm{Cl2}: \\
&\quad \text{if} \quad p_{Cl} > p_{Cl2} : p_b = p_{Cl2}; \quad T_b = T_{Cl2}; \quad p_a = p_{Cl}, \\
&\quad \text{if} \quad p_{Cl} > p_{Cl2} : p_b = p_{Cl}; \quad T_b = T_{Cl4}; \quad p_a = p_{Cl2}, \\
&\text{for } j = \mathit{Cl4}: \\
&\quad \text{if} \quad p_{Cl} > p_{Cl4} : p_b = p_{Cl}; \quad T_b = T_{Cl}; \quad p_a = p_{Cl4}, \\
&\quad \text{if} \quad p_{Cl4} > p_{Cl} : p_b = p_{Cl4}; \quad T_b = T_{Cl4}; \quad p_a = p_{Cl},
\end{aligned} \tag{45}$$

The cross-sections and the coefficients necessary for Eq. (9) are:

$$A_{Cl2} = A_{Cl4} = \pi d_{Cl}^2 n_{Cl}/4; \quad \zeta_{Cl2} = \zeta_{Cl4} = 0.85. \tag{46}$$

The approach applied to the cooler is identical to that one used for the heater and will not be repeated here. All the necessary parameters are determined from reasonable real values used in the simulation program.

3.4. Engine power and efficiency

The work of the expander and the compressor is calculated for the angle $0° \le \alpha \le 360°$, in the same way as for the 2-stroke engine:

$$L_E = \int_{\alpha=0^\circ}^{\alpha=360^\circ} p_E(\alpha) \frac{dV_E}{d\alpha} d\alpha, \tag{47}$$

$$L_C = \int_{\alpha=0^\circ}^{\alpha=360^\circ} p_C(\alpha) \frac{dV_C}{d\alpha} d\alpha. \tag{48}$$

The above given pressures are determined while taking into account mass losses in the valves.

The work of the engine is a sum of:

$$L = L_E + L_C - L_{B1} - L_{B2} - L_f, \tag{49}$$

where the values of work of the blowers are ($i = 1,2$):

$$L_{Bi} = 2\pi \dot{m}_{Bi}\ h_{SBi}/(\omega_{Bi} \eta_{Bi}). \tag{50}$$

The symbols used are explained in Appendix 1, and an enthalpy isentropic increase of blowers is:

$$h_{SBi} = \frac{\kappa}{\kappa - 1} R T_{bBi} \left(\Pi_{Bi}^{\frac{\kappa-1}{\kappa}} - 1 \right) \tag{51}$$

A complete power output of the externally heated 2-stroke engine is determined as:

$$P = \frac{\omega}{2\pi} L. \tag{52}$$

The heat delivered inside the heater (following the theoretical model description) is:

$$Q_H = \int_{\alpha=0^\circ}^{\alpha=360^\circ} \dot{Q}_{AH} \frac{d\alpha}{\omega}, \tag{53}$$

if the heat stream is calculated according to (27).

The theoretical and practical efficiencies of the engine are equal to:

$$\eta = L/Q_H, \quad \eta_{pr} = L/Q_{Hpr}. \tag{54}$$

The efficiency may be defined in a different way. Calculation of the value of Q_H is made without an additional heat exchanger. However, the temperature of the medium leaving the heater (Figs. 1 and 6) is very high, so a pre-heater should be applied like that one described in [10]. The pre-heater rises the temperature of a gas in front of the inlet to the heater and causes considerable savings of the heating medium (decreasing the denominator in Eq. (54) from Q_H to Q_{Hpr}). The efficiency of the EHVE, in this case, should increase significantly.

Table 1
Rotation and thermodynamic parameters of EHVE.

n_{EHVE} [rpm]	3000	2500	2000	1500	1000
T_{wH} max [K]	1273	1173	1073	973	873
ρ_{H1} [kg/m^3]	27.9	28.0	28.2	28.3	28.5
ρ_{Cl4} [kg/m^3]	14.0	14.1	14.3	14.5	14.9

4. Results of the computer simulations

The numerical task is reduced to a solution of the set of non-linear, ordinary differential equations (13), (15), (17), (19), (28), (30), (32), (34), (36), (38) with appropriate switching conditions, supplemented by a set of non-linear, algebraic equations. Stationary, periodic solution of this problem for engine operation at a given, constant angular velocity of the engine is presented. A solution of the custom written program built with the GCC compiler/linker is obtained using standard C++ language library procedures and solvers presented in [13]. The Runge-Kutta 4th order method is applied in the integration process. All simulations have started from identical, fixed initial conditions and continued until the solution reached integrated values differing in relative measures no more than 0.01% from those obtained at the end of the previous cycle. The numerical errors were within limits of accuracy of double precision numbers. Adaptive step-size integration method with forced decrease of step near switching values like valve opening/closing has been applied. The size of the integration step and limits of acceptance of the consecutive solutions has been chosen according to careful analysis of wide range of their values. It should be mentioned the system presents quick convergence of solution and rather small sensitivity to the step changes. Good agreement of the integrated values with experimental results published earlier in the paper [5] for the previous version of the engine makes the above statement satisfactory. Typically, stable solution has been reached after 100–150 cycles of simulated full crankshaft revolutions. The power of the described engine is calculated according to equations (49) and (52), the efficiency according to Eq. (54). The many of physical losses have been already taken into account. The heat transfer through cylinders and heat exchangers has not been considered. The losses caused by the heat transfer depend on a kind of insulation used and cannot be accurately defined at this stage. Therefore, it is estimated that total errors of the calculation in terms of effective power and efficiency can stay within 5% of results shown in Fig. 9.

The calculations were performed for $V_C/V_E = 0.75$, considered as the optimal ratio for the investigated case when $V_H \approx 5V_E$. The power and efficiency of the EHVE for this rate of the cylinder volumes are highest, but it is worth mentioning that both calculated characteristics are flat in this region. Similar optimization and determination of V_C/V_E is performed in [7], though physically both the blowers and their models introduced into the calculations are different. It is important that the above ratio is always less than 1 if we consider the engine with external heating.

Other essential data were:

$d_E = 0.083$ m, $s_E = 0.0834$ m,

$h_{0E} = 0.0025$ m – dimensions of the cylinder E.

$d_C = 0.083$ m, $s_C = 0.06255$ m,

$h_{0C} = 0.0017$ m – dimensions of the cylinder C.

$A_{1max} = A_{3max} = 0.001$ m^2, (two valves),

$A_{2max} = A_{4max} = 0.0005$ m^2 – valves surface areas.

Valves opening – index 0, closing – c, and respective transient angles – Δ:

$$\alpha_{10} = -10^\circ, \quad \alpha_{1c} = 80^\circ, \quad \Delta\alpha_1 = 35^\circ,$$
$$\alpha_{20} = 160^\circ, \quad \alpha_{2c} = 350^\circ, \quad \Delta\alpha_2 = 90^\circ,$$
$$\alpha_{30} = 110^\circ, \quad \alpha_{3c} = 190^\circ, \quad \Delta\alpha_3 = 35^\circ,$$
$$\alpha_{40} = 210^\circ, \quad \alpha_{4c} = 360^\circ, \quad \Delta\alpha_4 = 50^\circ,$$

The values of the above angles are established after preliminary optimization calculations.

Dimensions of the heat exchangers (H – heater, Cl – cooler), d – pipe diameter, l – pipe length, n – number of pipes, V – volume:

$$d_H = 0.01\,\text{m}, \quad l_H = 2.00\,\text{m}, \quad n_H = 15, \quad V_{H1} = V_{H3} = 0.005\,\text{m}^3,$$
$$d_{Cl} = 0.01\,\text{m}, \quad l_{Cl} = 2.00\,\text{m}, \quad n_{Cl} = 15, \quad V_{Cl2} = V_{Cl4} = 0.005\,\text{m}^3.$$

A value of the Δp_{H3} depends on the design of collectors and heat exchangers. For simplicity, all values of the Δp's are assumed to be 10 kPa, when the EHVE works at 3000 rpm.

$\Pi_{Bi} = 1.03$ – input to output pressure ratio in the blower.

T_{bBi} – temperatures taken from the outlet of the collectors H1 and Cl4.

$\eta_{Bi} = 0.7$ – assumed efficiency value of the blowers.

The temperature of the heater walls is assumed to decrease with the engine rotational speed and it is related to an amount of heating medium delivered, see Table 1. The minimum temperature of the cooler wall is constant and equal to $T_{wCl} = 300$ K, based on standard parameters of cooling water supply. The coefficients of the heat conduction, dynamic viscosity and specific heats are taken from the tables of physical properties of the air.

The electrical engines driving the blowers have to be controlled. Their temperatures and pressures at the inlet to the blowers, i.e., also the outlet collectors of the heat exchangers, decrease together with the rotation of the EHVE, but the density of the working air changes insensibly for different mass flow rates of blowers so the application of them is possible. The rotational speeds of the blowers are approximately constant and their angular velocities are $\omega_{Bi} = 314$ rad/s. The value of $\dot{m}_{Bi} \geq 0.06$ kg/s is the minimum blower mass stream (of given heat exchangers) for which Re $= 10^4$, when the valves of the engine are closed.

The rotation of the EHVE, the maximum temperature of its heater walls, the averaged densities of working air in the collectors H1 and Cl4 are shown in Table 1 for minimum mass flow rates of the blowers.

The program based on the described model can be used for different calculations. Figs. 5–8 show some examples, explained in detail in their respective captions with the blowers for $\dot{m}_{Bi} \geq 0.06$ kg/s.

The exception is Fig. 6, which will be described here. According to the theoretical model we show here a temperature of the expander T_E, a temperature of the compressor T_C, a temperature at the outlet from the heater T_H, a temperature at the outlet from the cooler T_{Cl}, and temperatures in exchanger collectors – T_{H1}, T_{H3}, T_{Cl2}, T_{Cl4}. The accuracy of the applied approach to the heat exchangers is checked, e.g., the pressure and the temperature at the outlet from the heater H are always close the pressure and the temperature in the outlet collector H1.

Fig. 9 shows power and efficiency of the engine with and without the blowers. The mass flow rate of both blowers are 0.1, 0.2 kg/s.

The heat delivered in presence of a pre-heater applied is smaller and is assumed to be $Q_{Hpr} = 0.85Q_H$. This assumption is rather modest and can be easily met in a real installation. The efficiency of the EHVE in this case is obviously higher than in case without presence of a pre-heater and is presented in Fig. 9 as η_{pr}.

5. Conclusions

A two-stroke externally heated valve engine (called EHVE), entirely different from the Stirling engine, has been investigated numerically. Two counter-current heat exchangers, a heater and a cooler are applied in this engine. The engine may be supplemented by two blowers (Figs, 1, 3 ,4), used to increase the volume of heat exchange when the valves of the EHVE are closed (Fig. 7).

A lack of the blowers has been also considered. The EHVE has the moderate exchangers (up to $5V_e$), works in a closed Joule cycle with air as a working medium, and additionally has a conventional crankshaft and an oil lubrication system.

A disadvantage of the EHVE realizing the Joule cycle is a short period of openings of valves Nos. 1 and 3, Fig. 7. It is a challenging but feasible engineering task. The statement is supported by the fact that intensive works connected with the valve timing are still developed, [15].

The calculations based on the model described are performed for $V_C/V_e = 0.75$, being the optimal value for which the EHVE power and efficiency are close to their optimum. Figs. 5–8 show pressures, temperatures, etc., described in their captions.

The diagram given in Fig. 9 is essential for this paper. It shows the efficiency and the power of the EHVE if blowers exist or not in the installation as functions of rotational frequency of the engine.

Taking into account that it is not an easy task to install the blower in the loop of high temperature and pressure, the profit of using blowers must be significant. In the present configuration, it is moderate, see Fig. 9.

The efficiency of the EHVE does not increase practically. The power of the engine for $\dot{m}_{Bi} = 0.1$ increases by about 25%. Further growth in the value of $\dot{m}_{Bi}$ has no practical sense, because the profit of using the blowers becomes insignificant.

Nevertheless, the EHVE without the blowers reaches an efficiency level of the same value as the Stirling engine, when a preheater is applied, like in [10].

Additionally, the EHVE uses atmospheric air while the Stirling one works with helium as the working gas [10]. Considering this, the power produced by 1 L of the EHVE at 1500 rpm, [10], when it works with the same gas, is calculated. The comparisons are following: the Stirling gives 51 kW, the EHVE without blowers – 59 kW. In this calculation, the conduction coefficients of helium are actually 5.3 times greater than these for the air.

The engine may be applied close to garbage dams which produce bio-gas, for propulsion of small boats or yachts when they are supplied by tanks of bio-gases. It may be also used on small nuclear submarines.

Figs. 5–8 show diagrams when the temperature of the heater varies in a range of 800–1000 °C. It is not always physically possible. For a solar heat source, so high temperature needs an application of a great set of special solar concentrators. The calculations of the engine for a lower temperature of the heater and rotational speed have been performed, but not published here. The engine operates properly in that case as well.

Ships of the future will be propelled by artificial sails but this fails when there is no wind and the sun shines intensively. The described engine may be a unique source of power in such a case.

Acknowledgement

Professor Lech Brzeski who died in 2006 was one of the main inventors of the EHVE. The authors of this paper would like to express their gratitude for his contribution.

Appendix 1

Notation

A	surface area [m^2]
C_v	specific heat at the constant volume [J/kg K]
D, d	diameters [m]
h	position of the piston [m]
h_j	valve lift [m]
h_{0E}, h_{0C}	clearances over the piston boundary positions for expander and compressor [m]
h_{SBi}	entalphy isentropic increase of blowers [J/kg]
L	mechanical work [J]
l	length of tubes [m]
$\dot{m}_j$	mass flow rates [kg/s]
p	pressure [N/m^2, bar]
P	power [W, kW]
Pr	Prandtl number [−]
n	number of tubes [−]
Q	heat [J]
$\dot{Q}_A$	heat flow in the surface area [W]
R	universal gas constant [J/kg K]
s	piston stroke [m]
Re	Reynolds number [−]
t	time [s]
T	temperature [K]
V	volume [m^3]
α	crankshaft angle [rad, grad]
α_A	heat transfer coefficient [W/m^2 K]
$\alpha_{j,k}$	angle of valves [rad]
$\Delta\alpha_j$	increase in a valve angle [rad]
ζ	loss coefficient [−]
η	efficiency [−]
κ	isentropic expansion index [−]
λ	heat conductivity coefficient [W/mK]
ρ	density [kg/m^3]
ω	angular velocity of the crankshaft [rad/s]

Subscripts

a	after
B1, B2	blowers
b	before
C	compressor
Cl	cooler
Cl2, Cl4	collectors of the cooler
E	expander
f	friction
H	heater
H1, H3	collector of the heater
i	number of blowers (=1,2)
j	number of a valve (=1, 2, 3, 4)
k	opening ($k = 0$), closing ($k = c$)
max	maximum
w	wall

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